AEDC-TMR-90-E14

C#1

DIGITAL SIMULATION AND MODELING OF A TURBINE ENGINE HORSEPOWER EXTRACTION SYSTEM

Ronald T. Bishel
Turbine Engine Division
Directorate of Propulsion Test
Arnold Engineering Development Center

August 1990
Final Report for Period January - May 1990

Approved for public release; distribution is unlimited.

ARNOLD ENGINEERING DEVELOPMENT CENTER
ARNOLD AIR FORCE BASE, TENNESSEE
AIR FORCE SYSTEMS COMMAND
UNITED STATES AIR FORCE

Report Documentation Page				*Form Approved OMB No. 0704-0188*
Public reporting burden for the collection of information is estimated to average 1 hour per response, including the time for reviewing instructions, searching existing data sources, gathering and maintaining the data needed, and completing and reviewing the collection of information. Send comments regarding this burden estimate or any other aspect of this collection of information, including suggestions for reducing this burden, to Washington Headquarters Services, Directorate for Information Operations and Reports, 1215 Jefferson Davis Highway, Suite 1204, Arlington VA 22202-4302. Respondents should be aware that notwithstanding any other provision of law, no person shall be subject to a penalty for failing to comply with a collection of information if it does not display a currently valid OMB control number.				
1. REPORT DATE **AUG 1990**	2. REPORT TYPE			3. DATES COVERED **00-01-1990 to 00-05-1990**
4. TITLE AND SUBTITLE **Digital Simulation And Modeling Of A Turbine Engine Horsepower Extraction System**				5a. CONTRACT NUMBER 5b. GRANT NUMBER 5c. PROGRAM ELEMENT NUMBER
6. AUTHOR(S)				5d. PROJECT NUMBER 5e. TASK NUMBER 5f. WORK UNIT NUMBER
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) **Arnold Engineering Oevelopment Center,Turbine Engine Division,Directorate of Propulsion Test,Arnold Air Force Station,TN,37389**				8. PERFORMING ORGANIZATION REPORT NUMBER
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)				10. SPONSOR/MONITOR'S ACRONYM(S) 11. SPONSOR/MONITOR'S REPORT NUMBER(S)
12. DISTRIBUTION/AVAILABILITY STATEMENT **Approved for public release; distribution unlimited**				
13. SUPPLEMENTARY NOTES				
14. ABSTRACT **see report**				
15. SUBJECT TERMS				
16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
a. REPORT **unclassified**	b. ABSTRACT **unclassified**	c. THIS PAGE **unclassified**	**Same as Report (SAR)**	**54**	

Standard Form 298 (Rev. 8-98)
Prescribed by ANSI Std Z39-18

APPROVAL STATEMENT

This report has been reviewed and approved.

Ronald T. Bishel

RONALD T. BISHEL, GS-12
Turbine Engine Division
Directorate of Propulsion Test
Deputy for Operations

Approved for publication:

FOR THE COMMANDER

Richard F. Austin

RICHARD F. AUSTIN
Deputy Director of Propulsion Test
Deputy for Operations

ABSTRACT

Digital simulation and modeling of a turbine engine horsepower extraction system is presented. The system consists of a digital valve, a fluid transmission line (piping), and a waterbrake. The system model was developed using the basic lumped-parameter approach. A computer modeling language called Advanced Continuous Simulation Language (ACSL) represents the model's response to step and profile inputs. The simulation of step and profile inputs demonstrate the control system's ability to set power levels and control changes much faster and with better accuracy than a manual system. The waterbrake equation was empirically derived and validated.

TABLE OF CONTENTS

CHAPTER PAGE

LIST OF FIGURES

NOMENCLATURE

ACSL	Advanced Continuous Simulation Language
AEDC	Arnold Engineering Development Center
A	Area ($in.^2$)
B	Bulk Modulus ($lbf/in.^2$)
CE	Equivalent Capacitance (lbm)/($lbf/in.^2$)
CF	Conversion Factor
CHP	Change Water Flow/Horsepower (lbm/(sec hp))
CV	Valve Flow Coefficient
D	Pipe Diameter (in.)
dp	Differential Pressure (psi)
dt	Change in Time (seconds)
F	Force (lbf)
f	Friction Factor
gc	Gravitational constant (lbm ft/lbf sec^2)
HPX	Horsepower Extraction (hp)
HPA	Actual Horsepower (hp)
HPD	DesiredHorsepower (hp)
HPE	Error Horsepower (hp)
L	Length (ft)
LE	Equivalent Inductance ($lbf/in.^2$/(sec^2/lbm))
N	Engine Rotor speed (rpm)
P	Pressure (psi)
psig	Gauge pressure ($lbf/in.^2$)
r	Radius (in.)

RE	Equivalent Resistance ($lbf/in.^2/(lbm^2/sec^2)$)
RN	Reynolds Number
R_W	Waterbrake Flow Resistance
T	Torque (lbf ft)
t	Time (seconds)
$\dot{w}$	Mass Flow Rate (lbm/sec)
V	Volume (ft)
v	Velocity (ft/sec)
ν	Voltage
θ	Angular Velocity (rad/sec)
μ	Viscosity (lbm/ft sec)
ρ	Density (lbm/ft)
e	Pipe Roughness (in.)
$\int$	Integral Sign

Subscripts

1	Valve
2	Transmission
3	Waterbrake
a	Air
h	Holes
g	Gap
s	Shaft
t	Total
w	Water

Introduction

Modern control systems are necessary to control complex tasks effectively. A human operator is unable to correct for error in milliseconds. A digital computer can accomplish complex tasks in milliseconds. Many of today's computer controllers provide reliable and consistent error corrections on both complex and repetitive tasks. To control a system effectively, whether a heating system or rocket guidance system, the system must be understood. Modeling subdivides a complex system into its basic components. Once a model is developed, whether digital or analog, one needs to evaluate the control's ability to regulate the system.

This thesis models a gas turbine horsepower extraction system (waterbrake) using a computer modeling language called Advanced Continuous Simulation Language (ACSL). The control requirements for a waterbrake system are stated in Chapter I. Chapter II develops the horsepower extraction (HPX) model from the basic parameters and system equations for each component. The computer program, written in ACSL, solves the system equations in Chapter III. The model response to various inputs - step and profile inputs are shown in Chapter IV. Chapter V summarizes the results and provides a recommendation.

CHAPTER I

CONTROL REQUIREMENTS

Background

Most of the turbine engine horsepower extraction (HPX) systems at AEDC use a waterbrake connected to the engine gearbox shaft to simulate aircraft accessory loads. Presently the HPX is set manually which is time consuming and does not provide accurate simulation of aircraft accessory loads during transient engine operations. Setting the HPX automatically would save a few minutes during each airstart test. Testing turbine engines at Arnold Engineering Development Center (AEDC) is expensive and a few minutes saved each test applied to the total number of tests means a potential saving (for thirty tests) of approximately $35,000.00/year. Presently, the HPX is set at one engine shaft speed (rpm). This setting is not adjusted for changes in shaft speed. As a result, the horsepower extracted does not match the typical aircraft accessory loads. A computer controller can automatically correct for changes in shaft speed and adjust water flow to the waterbrake during engine simulated airstarts. Setting the HPX level correctly and continuously would significantly reduce the difference between test simulated HPX and actual aircraft HPX (Figure 1).

Waterbrake

The waterbrake is a key component in the HPX system. The waterbrake is a power absorption device driven by a shaft from the engine's gearbox (Figure 2). Power is absorbed by water as it passes

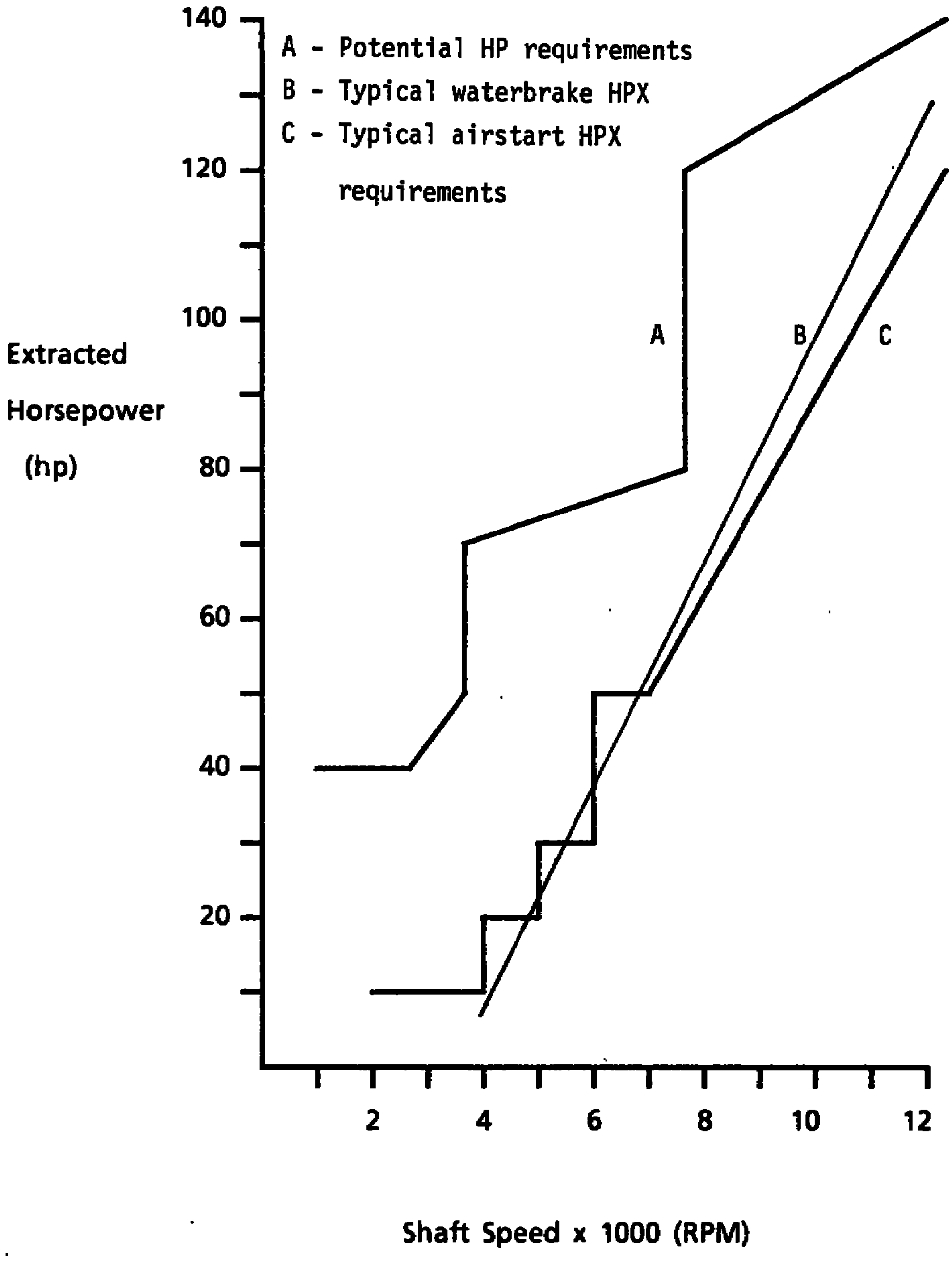


Figure 1. Typical Aircraft Horsepower Loads Versus Simulated Loads

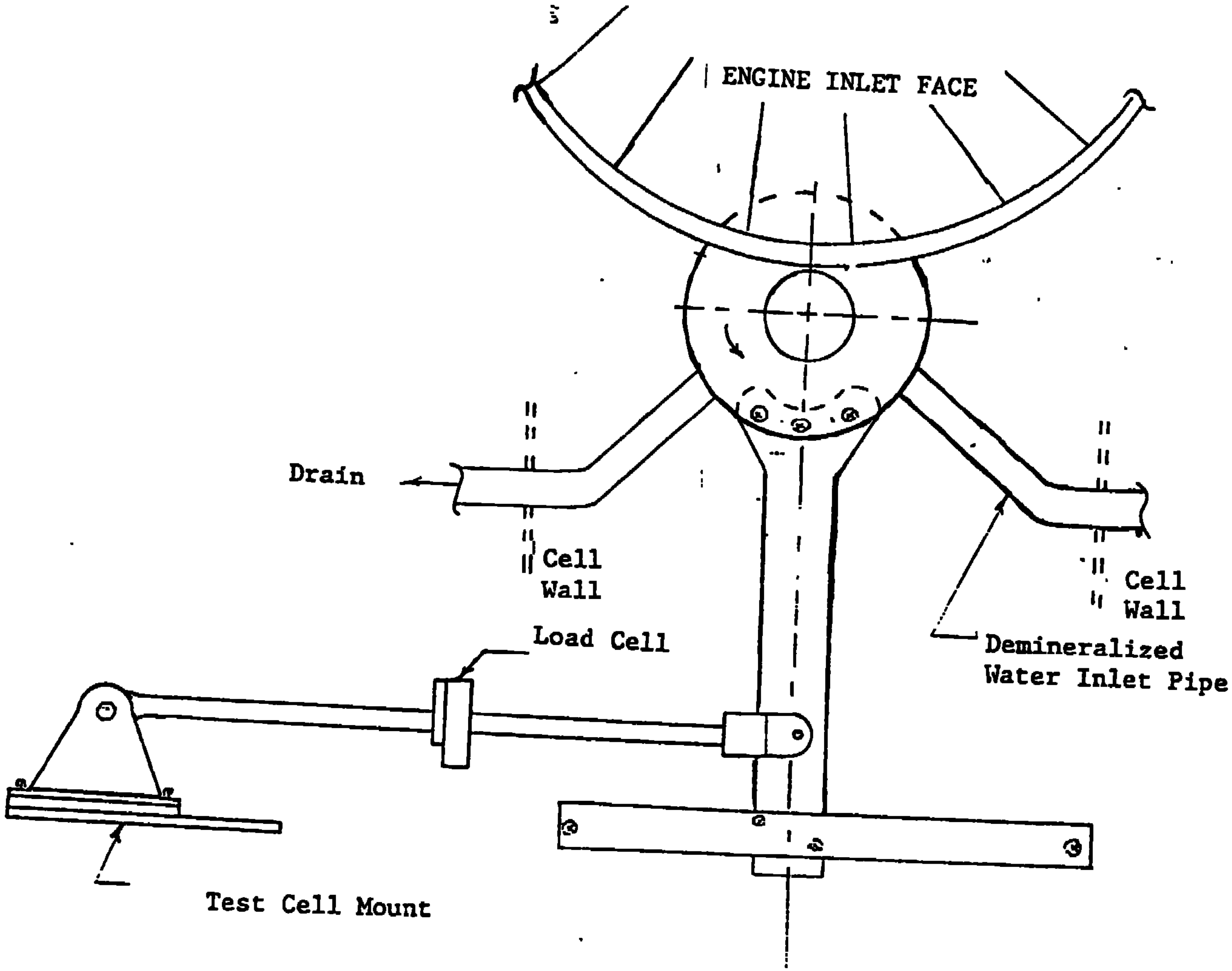


Figure 2. Waterbrake System Hardware

through a series of rotating discs and stationary plates. The water is discharged from the waterbrake at a higher temperature due to the basic energy absorption process. The level of power absorbed by the waterbrake can be regulated by controlling the water flow rate or shaft speed [1][1].

The waterbrake horsepower extraction system uses reaction torque and rotor speed measurements to determine horsepower. Measuring the torque and the speed provides almost instantaneous readout capability. The waterbrake system hardware shown in Figure 2 uses a trunion bearing, a lever arm, and a load cell to measure torque. The engine's gearbox shaft speed is determined from the engine core speed pickup and a known gearbox ratio. The correct size of waterbrake required is determined from the brake's power absorption capacity curves. Figure 3 shows the maximum HPX versus a range of shaft speeds (rpm) for typical waterbrakes used at AEDC.

System Operation

The current horsepower extraction system is manually operated as shown on the top of Figure 4. A one-inch water supply line feeds water at 65 psig, into a control valve. A 0.5-inch pipe transports the water to the waterbrake 15.5 feet away; where it passes through the waterbrake and is drained. A load cell measures the reaction force using a trunion bearing and linkages (Figure 2). The extracted

---

[1]Numbers in brackets refer to similarly numbered references in the List of References.

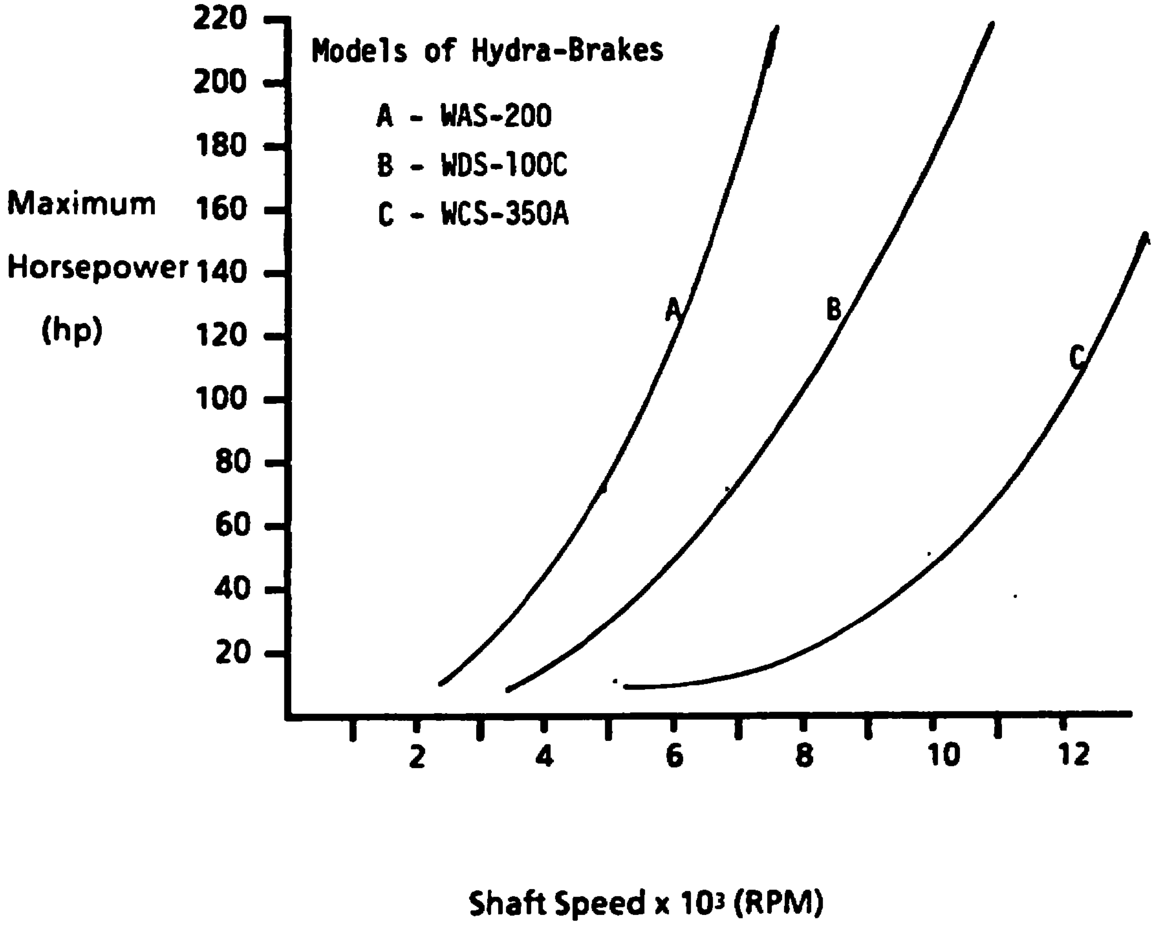


Figure 3. Waterbrake (Hydra-brake) Horsepower Curves

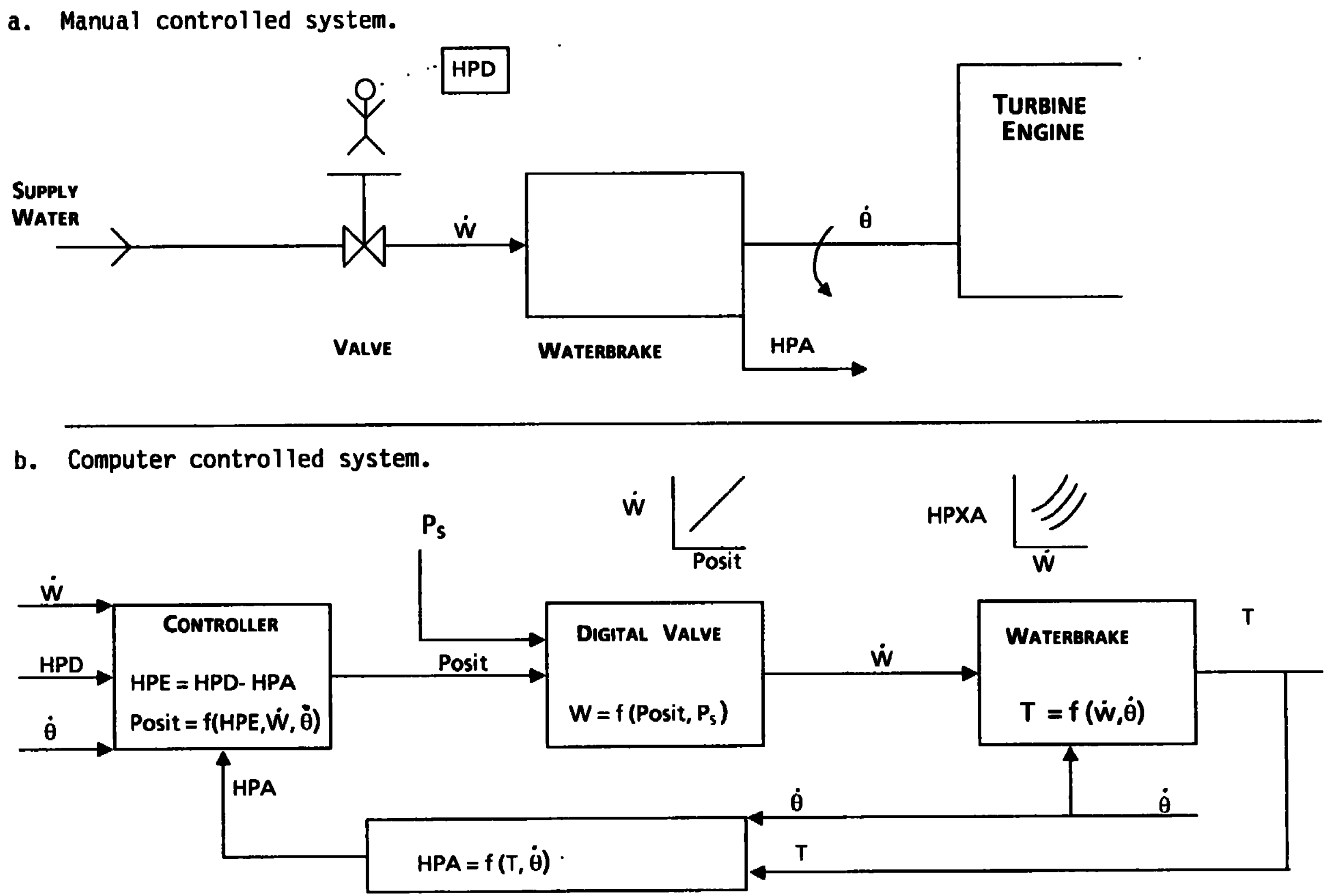


Figure 4. Comparison Between Manual and Computer Controlled Systems

horsepower is calculated from the reaction force, linkage length, and gearbox shaft speed. The automatic control system on the bottom of Figure 4 constantly corrects to the desired horsepower by using a computer program instead of an operator to calculate power and automatically adjust water flow.

CHAPTER II

MODELING THE HORSEPOWER EXTRACTION SYSTEM

Control System

The primary control system goal is to set the desired horsepower accurately during steady-state engine operations which saves expensive engine testing time. The secondary goal is to provide relative control during an engine air start transient characterized by rapid rotor speed increase and step increases in horsepower demands. A mathematical model was written representing the physical system analytically, using a series of equations. A computer model simulation will help determine if the physical system can meet the primary and secondary goals mentioned above.

The horsepower extraction (HPX) control system has four main components: control valve, transmission line, waterbrake and controller. Each one of the main components can in turn be represented by a lumped-parameter model.

Modeling Technique

The lumped-parameter technique assumes the physical distributed characteristics of pipe wall friction (resistance, RE), fluid compressibility (capacitance, CE), and water momentum (inductance, LE). along the pipe can be estimated by one lump (value) for each of the three characteristics (RE, CE, LE). By reducing the distributed characteristics of a pipe into lumped-parameters (RE, CE, LE) the complex and cumbersome equations of the flow can be simplified.

Further information on this topic can be found in Reference [2]. The lumped-parameter approach reduces the fluid (physical) system into its basic fluid parameters. The correlation between the fluid terms and electrical terms are shown in Table 1. The three basic fluid parameters, equivalent resistance (RE), equivalent capacitance (CE), and equivalent inductance (LE), are determined for the valve, transmission line, and the waterbrake.

The fluid equations and basic parameters are developed for the transmission line on the next few pages.

Equivalent Resistance (RE)

Calculation of equivalent resistance (RE) shown in Figure 5 is computed from Bernoulli's Equation [3].

$$H = f\left(\frac{L}{D}\right)\frac{v^2}{2gc} = \frac{dp}{\rho} \tag{1}$$

where

H = head loss (ft lbf/lbm)	D = pipe diameter in inches
f = friction factor	v = fluid velocity velocity in ft/sec
L = pipe length in inches	gc = 32.2 ft lb_m/lb_f sec^2
ρ = density in lb_m/ft^3	dp = change in pressure (lb_f/ft^2)

Table 1. Correlation of Mechanical and Electrical Terms

Terms	Electrical	Fluid Equivalent	
		Symbols	Description
Resistance	R	RE	fL/(D 2g p A2)
Capacitance	C	CE	$\rho V/B$
Inductance	L	LE	L/A gc
Voltage/Pressure	v	P	pressure (p)
Current/Mass Flow	i	$\dot{w}$	Mass Flow (w)
Voltage Drop/ Pressure Drop	dv	dp	pressure drop

where

L = pipe length

D = pipe diameter

ρ = water density

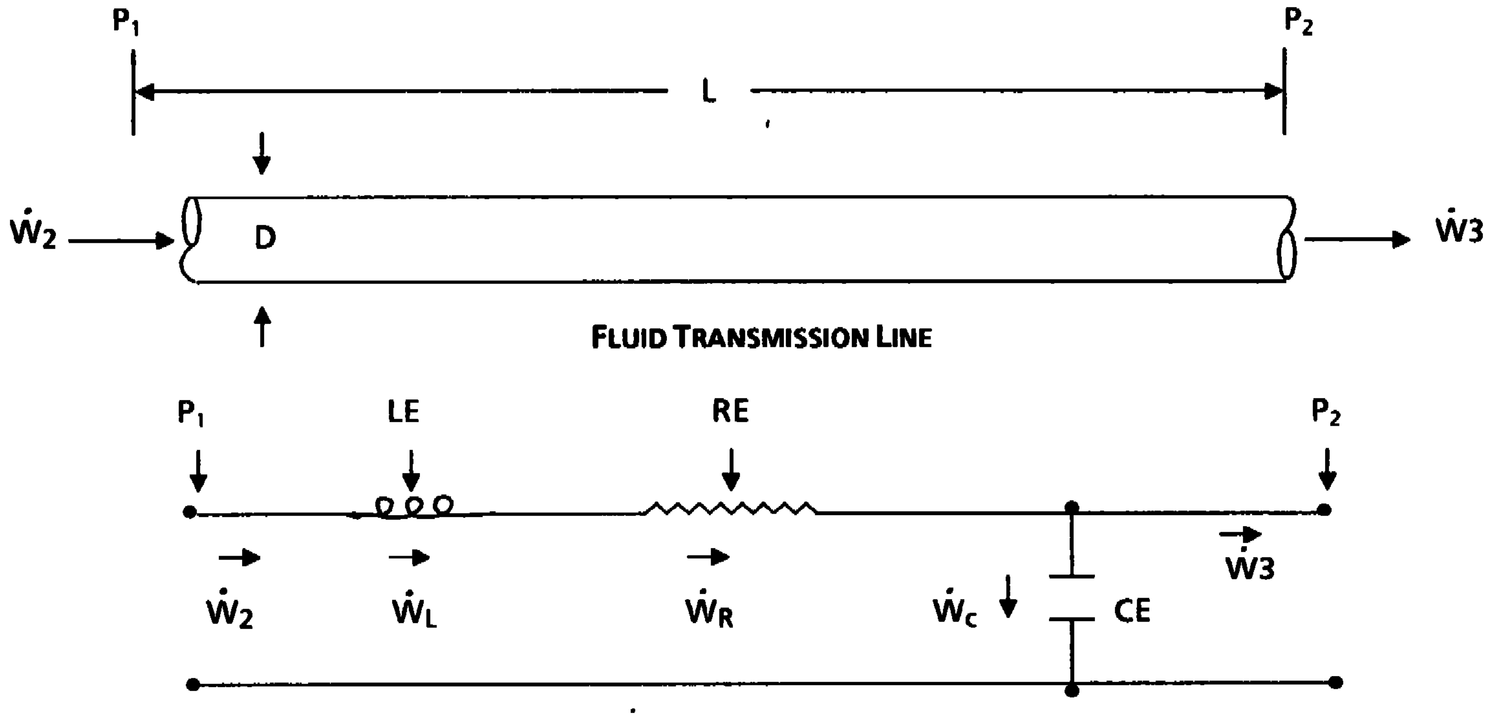
A = pipe's cross-sectional area

gc = conversion factor

f = frictional factor

B = bulk modulus

V = volume



P_1 and P_2 = lbf/in^2 Analogous to Voltage

w_2 and W_3 = lb$_m$/sec Analogous to Current

$W_L = W_R = W_2$ = Flow Through Inductor and Resistor

$W_c = W_3 - W_2$ = Flow Through Capacitor

Figure 5. Fluid to Electrical Transmission Line Analogy

$$H = \frac{P_1}{\rho} - \frac{P_2}{\rho} \tag{2}$$

and

$$P_1 - P_2 = \rho f\left(\frac{L}{D}\right)\frac{v^2}{2gc}. \tag{3}$$

However,

$$v = \left(\frac{w}{\rho A}\right), \textit{ where } w = \textit{flow rate lbm/sec}, \tag{4}$$

and

$$P_1 - P_2 = \rho f\left(\frac{L}{D}\right)\frac{\dot{w}^2}{\rho^2 A^2 2gc}. \tag{5}$$

If we let

$$RE = \frac{f\left(\frac{L}{D}\right)}{\rho A^2 2gc}. \tag{6}$$

The equivalent resistance (RE), of the transmission line is calculated below.

$$RE_2 = \frac{fL_2}{D_2 \rho_w A_2^2 2gc} \tag{7}$$

where

L_2 = effective length of transmission line pipe = 15.5 feet

D_2 = internal pipe diameter = 0.5 inch

ρ_w = water density = 62.4 lbm/ft^3

A_2 = cross-sectional area of the pipe = $D_2^2/4$

A_2 = 0.196 Sq inches = 1.36 E-3 Sq feet

g_c = gravitational constant = 32.2 lbm ft/lbf sec

f = frictional factor.

To determine the frictional factor (f) one needs the Reynolds Number (RN) and the pipe roughness e/D [4].

e = roughness, commercial = 0.00015

e/D_2 = 0.00003

$RN = \rho_w \; V_2 \; D_2/(\mu_w \, g_c)$ (8)

μ_w = absolute viscosity of water @ T = 68°F

= 2.09 E-5 lbf sec /sq ft

V_2 = average velocity = average volume flow/A_2 (9)

= 8 gal/min (ft^3/7.48 gal) (min/60 sec)/1.36 E-3 ft

= 12.96 ft/sec

RN = 4.98 E 4 (turbulent flow)

f = 0.022 from Moody chart [3].

Knowing the frictional factor we can now calculate the equivalent resistance (RE_2),

RE_2 = 7.61 (lbf/in^2)/(lbm^2/sec^2).

Equivalent Capacitance (CE)

The next constant to be determined is the capacitance. The line capacitance is attributed to the compression of the fluid (water), compression of the entrained air and the expansion of the pipe. Since the pressure in the pipe is very low (less than 65 psi), the pipe expansion can be neglected. The amount of air entrained in the water is not

known but considered to be between 0 to a few percent. Some air entrainment is typical in a water system and one percent by volume was used in the simulation model. The equivalent capacitance (CE) in Figure 5, page 12, can be computed by considering the Bulk Modulus (B).

$$CE_2 = \frac{\rho_w A_2 L_2}{B} \tag{10}$$

where B = bulk modulus water with entrained air one percent by volume

1/B = 1/Bw +1/Ba [2]

1/B = (0.99/3.25E5 + 0.01/65)

B = 6374 lbf/sq in.

Therefore,

CE_2 = 2.13 E-4 lbm sq in/lbf

The equivalent capacitance (CE) is derived below:

$$B = \frac{dp}{dv} V \tag{11}$$

and

$$V = volume\, ft^3$$

$$dV = \frac{dp}{B} V$$

$$dp = \frac{B}{V} dV$$

therefore,

$$\frac{dp}{dt} = \frac{B}{V} \frac{dV}{dt}. \tag{12}$$

However

$$\frac{dV}{dt} = \frac{\dot{w}}{\rho}$$

$$\frac{dp}{dt} = \frac{B}{V\rho}\,\dot{w}. \tag{13}$$

Defining

$$CE = \frac{V\rho}{B} \tag{14}$$

then

$$p = \frac{1}{CE}\int_0^t \dot{w}\,dt. \tag{15}$$

Equivalent Inductance (LE2)

Inductance (Figure 5, page 12) can be calculated by considering Newton's second law [2, 5], and assuming RE = 0, then

$$F = \frac{ma}{g_c} \tag{16}$$

where,

g_c = 32.2 ft lbm/lbf sec^2,

F = force, lbf,

m = mass of fluid lbm,

a = acceleration of the fluid in ft/sec^2

then

$$F = \Delta PA \tag{17}$$

where

$$\Delta P = P_1 - P_2.$$

The mass of fluid (m) in the line of length L is,

$$m = \rho AL. \tag{18}$$

Let

$$acceleration\; a = \frac{dv}{dt} \tag{19}$$

then

$$F = \frac{ma}{g_c} = \Delta PA = \frac{\rho AL}{g_c}\frac{dv}{dt}. \tag{20}$$

Define

$$v = \frac{\dot{w}}{\rho A}, and\; \frac{dv}{dt} = \frac{1}{\rho A}\frac{d\dot{w}}{dt} \tag{21}$$

then

$$\Delta PA = \frac{\rho AL}{g_c}\frac{1}{\rho A}\frac{d\dot{w}}{dt}$$

therefore

$$\Delta P = \frac{L}{g_c A}\frac{d\dot{w}}{dt}. \tag{22}$$

Let

$$LE = \frac{L_2}{g_c A_2}, \tag{23}$$

where L_2 = pipe's length = 15.5 feet

g_c = conversion factor

A_2 = pipe's cross-sectional area

= 0.196 square inches.

Fluid Transmission Line

The fluid transmission line consists of five sections of pipe, and three 90 elbows which transport the water from the digital valve to the waterbrake. The fluid to electrical transmission line analogies are presented in Figure 5, page 12, and the description of the equations are shown below.

The fluid transmission line is converted to a electrical circuit to model a control system. The same techniques used to solve an electrical circuit will apply to the analogous fluid circuit. The pressure drop (analogous to voltage) drop from P_1 to P_2 is equal to pressure drop across the inductor (LE) plus the pressure drop across the resistor (RE) [2,5,6].

$$P_1 - P_2 = LE_2 \frac{d\dot{w}}{dt} + RE_2 \dot{w}^2 \tag{24}$$

Electrically this would be analogous to

$$v_1 - v_2 = L \frac{di}{dt} + R\,i \tag{25}$$

solving for $\dot{w}_2$

$$\dot{w}_2 = \frac{1}{LE_2} \int_o^t \left[P_1 - P_2 - RE_2 (\dot{w}_2)^2 \right] dt \tag{26}$$

The pressure P_2 is defined by the Equation (28) using the fact from the law of conservation of matter.

Water stored equals water in minus water out is shown in Equation (27) below.

$$CE_2 \frac{dP}{dt} = \dot{w}_1 - \dot{w}_2 \tag{27}$$

Rearranging terms to solve for P_2

$$P_2 = \frac{1}{CE_2} \int_o^t (\dot{w}_1 - \dot{w}_2) dt \tag{28}$$

<u>Waterbrake Power Equation</u>

The waterbrake is a compact power absorption device used to simulate loads on turbines and other rotating equipment. Equation (32) below calculates the horsepower absorbed assuming the power is dissipated by acceleration and deceleration of the fluid. Figure 6 shows a cross-sectional view of the waterbrake with the radius to the shaft, gap, and holes.

$$\text{Power} = \text{Torque x Angular Velocity} = T \times \theta \tag{29}$$

and

$$\text{Torque} = \text{force x distance} = F \times r \tag{30}$$

since

$$\text{Force} = \text{mass x acceleration} = \dot{w}\ dv/dt = \dot{w}\ \theta\ r \tag{31}$$

gives us

$$T = CF_w\ \dot{w}\ \theta r^2$$

The total torque equals rotor torque from the water passing ($\dot{w}_h$) through the holes (r_h) plus water passing (w_g) through the gap (r_g)

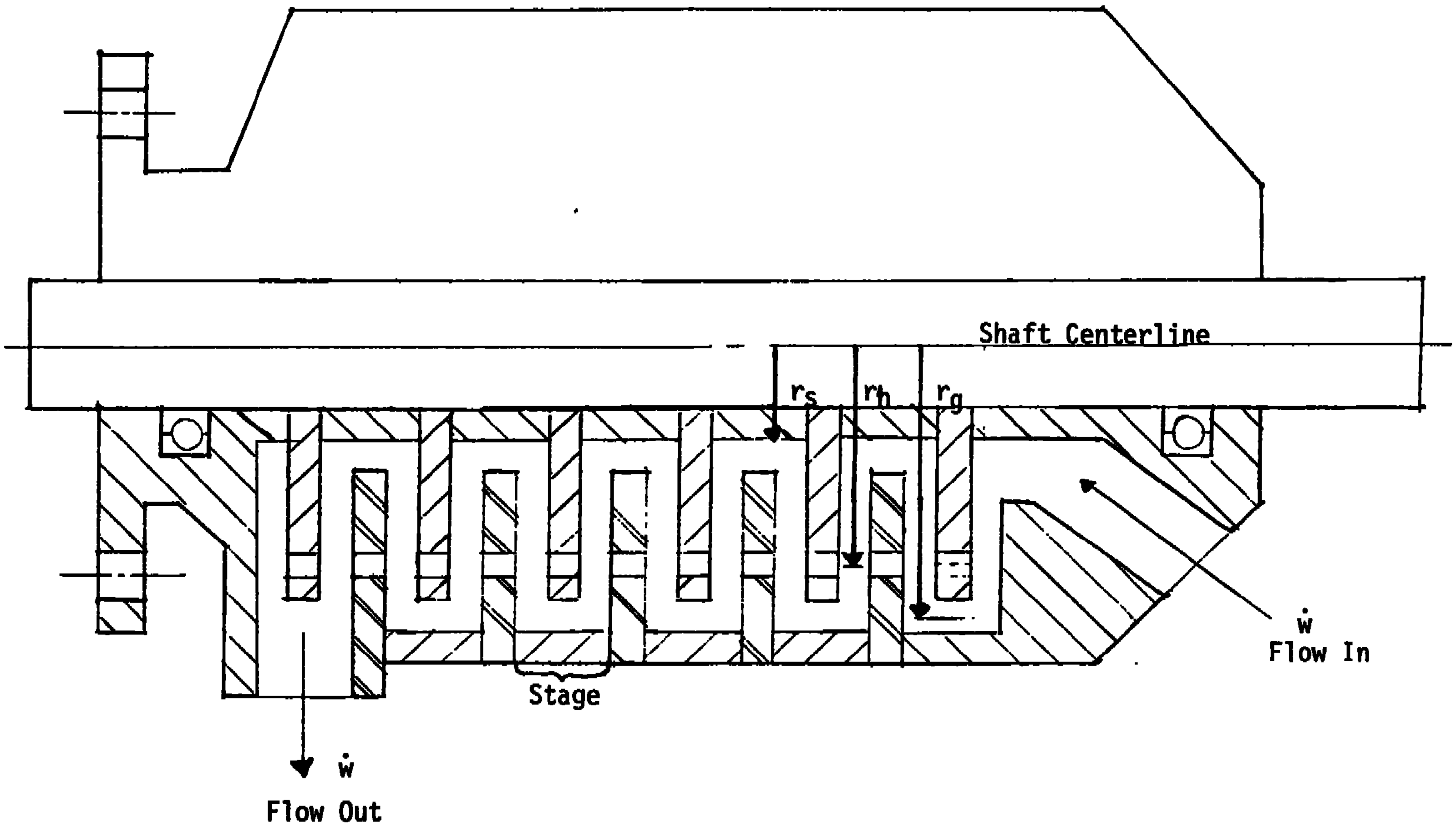


Figure 6. Cross-Sectional View of the Waterbrake

minus initial torque from the water passing over rotor shaft (r_s). This equation is written out below for six stages:

$$T = CF_w \{\dot{w}_h r_h \ (r_h \ \theta) + w_g r_g (r_g \theta)$$
$$- \dot{w}_s r_s \times (r_s \ \theta)\} \times 2/\text{stage (6 stages)} \qquad (32)$$

where

r_h = radius from shaft centerline to the holes (2.50 inches)

r_g = radius from shaft centerline to the gap (3.00 inches)

r_s = radius of the shaft (1.00 inches)

stage = rotor + stator (shown in Figure 6)

θ = rotor speed (N) x 1.144 (gearbox multiplier), rpm

$\dot{w}$ = water flow in gal/min

$\dot{w}_t = \dot{w}_s$ = total water flow

$\dot{w}_s$ = water flow around the shaft at radius r_s

$\dot{w}_h$ = water flow through the holes at radius r_h

$\dot{w}_g$ = water flow through the gaps at radius r_g

The total flow through the waterbrake is equal to the flow through the hole plus the flow through the gap as written in Equation (33)

$$\dot{w}_T = \dot{w}_h + \dot{w}_g \qquad (33)$$

Let $W_T = x \dot{w} + y \dot{w}$

where x = percentage of total water through the rotor holes

y = percentage of total water through the rotor gap

Based on flow area of the rotor holes and the circumferential gap, and the equation fit to the waterbrake data. The x and y values were chosen to be: x = 0.20 and y = 0.80.

Factoring out θ, gives the torque equation

$$T = CF_w \dot{w} \ \theta(xr_h^2 + yr_g^2 - r_s^2) \tag{34}$$

The waterbrake horsepower can now be determined from the power equation shown below.

$$\text{Power} = [CF_p \ \theta^2 \dot{w} \ (x \ r_h^2 + y \ r_g^2 - r_s^2)] \ \ 6 \text{ stages} \tag{35}$$

CF_p = conversion factor = 5.99 E-10.

Table 2 compares the experimental test data to the analytical HPX values for various water flow rates and rotor speeds. The comparison is incomplete, due to the lack of test data.

Waterbrake Flow Rate Equation

The flow rate ($\dot{W}$) is very complex and not easily modeled. For this reason, an empirical equation was derived from the test data available. The flow equation is as follows:

$$\dot{w} = (dP/RE_3)^{1/2} \tag{36}$$

where

$\dot{w}$ = water flow through the waterbrake,

dP = pressure across the waterbrake,

RE_3 = resistance which is a function of both inlet pressure, rotor speed and flow rate.

Data used to determine RE_3 can be found in Figure 7. The inductance (LE_3) and capacitance (CE_3) waterbrake parameters were not considered due to lack of dynamic test data to characterize waterbrake.

Table 2. Actual Waterbrake Data Versus Analytical Horsepower Values

Rotor Speed (N) rpm	Flow Rate, $\dot{W}$ lbm/sec	Equation Output, HPX hp	Experimental Data, HPX hp	Different Equation-Exp. HPX hp
13,000	2.22	189	190	-1
12,500	2.09	164	180	-16
11,800	0.76	54	80	-26
11,800	1.25	88	110	-22
10,000	0.70	39	45	-6
2,500	1.95	6	10	-4

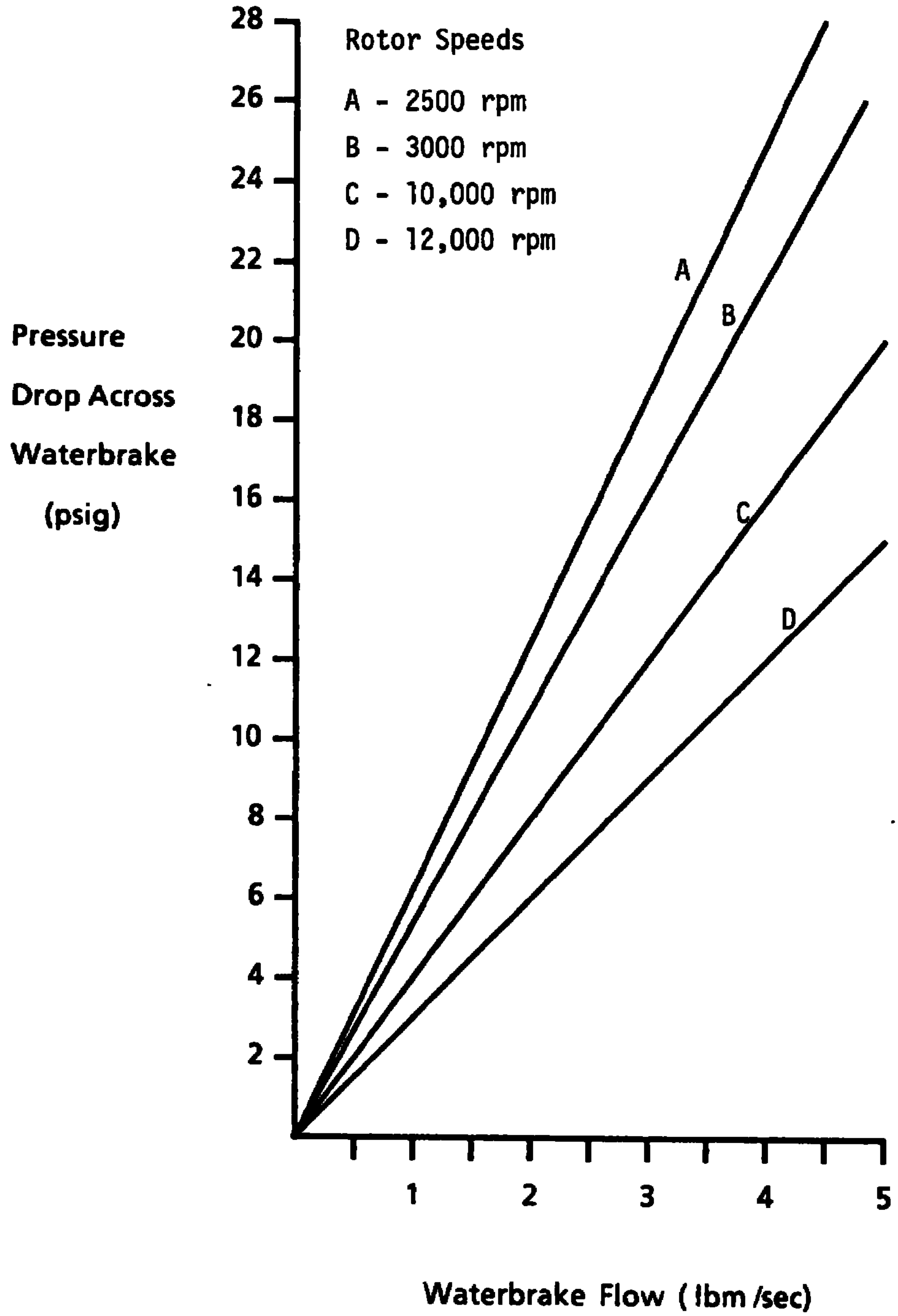


Figure 7. Empirical Waterbrake Data with Lines of Constant Resistance at Designated Rotor Speeds

Digital Control Valve

A digital twelve element control valve, manufactured by The Digital Control Company [7], was chosen to regulate the water flow. The high-tech valve has a (full travel) response time of less than 100 milliseconds, no hystersis, and a linear CV trim. The general form of the control valve equation is shown below [6].

$$\dot{w} = CF_v CV\left(\sqrt{dP}\right) \tag{37}$$

where

dP = pressure across the valve (psig)

CV = flow coefficient defined as the volume of water in gallon/minute that will flow with a one psi pressure drop

CF_V = conversion factor.

The digital valve was sized for a maximum water flow of 16 gpm, a 9 psig across the valve, and a flow resolution of 0.015 gpm. The twelve elements (solenoids) each have separate orifices that act as switches when powered by 100 psig.

The capacitance for the 1.0-inch valve that is 12 inches long can be found in Table 3. The valve inductance of (0.04) is neglected compared to the transmission line inductance of (2.46), therefore, and will be ignored to reduce the complexity of the control system simulation.

Table 3. Summary of Calculated Parameters

Section A: Valve

$RE_1 = f$ (valve area, dp)

$LE_1 = 0.040$ (lbf/in.^{2})/(lbm/sec^2)

$CE_1 = 5.23$ E-5 (lbm)/(lbf/in.^{2})

Section B: Transmission Line

$RE_2 = 7.61$ (lbf/in.^{2})/(lbm^2/sec^2)

$LE_2 = 2.46$ (lbf/in.^{2})/(lbm/sec^2)

$CE_2 = 2.13$ E-4 (lbm)/(lbf/in.^{2})

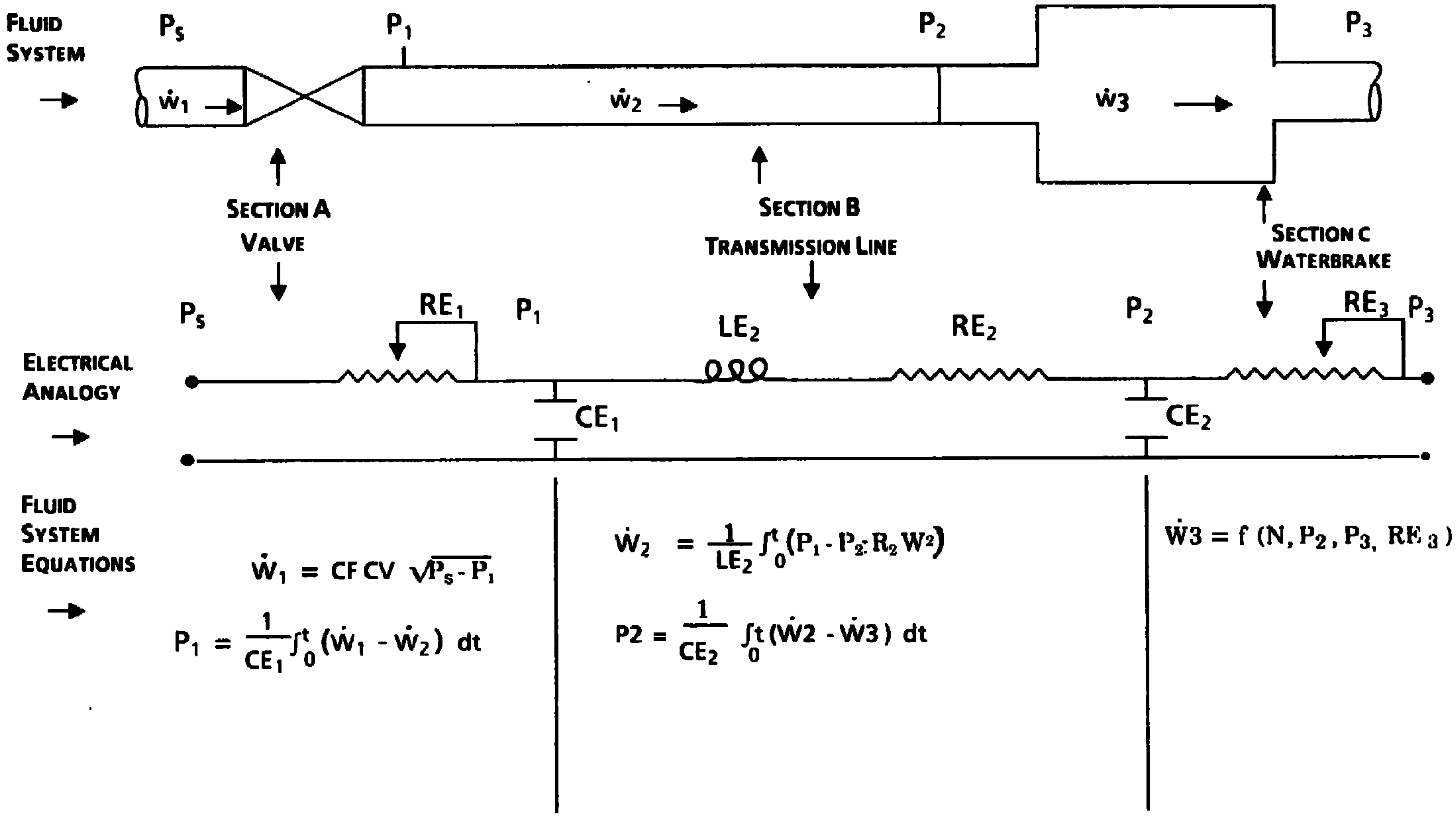
Section C: Waterbrake

$RE_3 = f(dp, \theta, \dot{w})$

$\left.\begin{array}{l} LE_3 \\ CE_3 \end{array}\right\}$ Not considered

System Equations

The system equations and a flow schematic of the digital valve, transmission line and waterbrake are shown on Figure 8. The electrical electrical analogy of the mechanical system is also shown on Figure 8. Summary of the calculated parameters (CE, RE, and LE) are provided in Table 3. Equations (7), (10), and (23) define the mechanical equivalent resistance (RE), capacitance (CE), and inductance (LE). The transmission line Equations (26) and (28) define the flow ($\dot{w}$) and pressure (P) terms. The waterbrake flow rate ($\dot{w}_3$) Equation (36) was derived empirically from the experimental data. The digital valve water flow ($\dot{w}_1$) and downstream pressure (P_2) were determined by Equations (26) and (28), respectively.



NOTE: RE_2, LE_2, CE_2 and CE_1 were determined by equations 7, 23 and 10.
The Equations for W_1, W_2 and W_3 are equations 37, 26 and 36.
The Equations P_1 and P_2 are both from equation 28.

Figure 8. Horsepower Extraction (HPX) Parameters, Locations, and System Equations

CHAPTER III

COMPUTER SIMULATION PROGRAM

Background

The system equations can be solved by several approaches. Some approaches are: analog computer simulation, custom numerical computer simulation, or Advance Continuous Simulation Language (ACSL). The extreme flexibility and computing power of ACSL make it the most attractive approach. The highlights of the program are: uses conventional Fortran statements, has nonlinear function generation up to three variables and flexibility to plot model behavior with various forcing functions. The ACSL manual [8] provides the detail of its capabilities.

Computer Program

The system equations, shown in Figure 8 were programed using ACSL. The computer program is given in Table 4. The program is separated into four sections: valve, transmission line, waterbrake and controller. The flow chart (Figure 9) and system equations (Figure 8) show the interdependence of each component. The beginning of the program provides the initial conditions, the parameter constants (CE, RE, LE), and the look-up tables of nonlinear data (shown graphically in Figures 10-13). Figure 11 shows at lower rotor speeds a greater change in water flow is required to obtain the same change in the amount of horsepower. The physical system is bound between 0.05-65 psi and 0.01 (leakage) to 2.30 lbm/sec. The valve position range is 0.1 to 98 percent of full

Table 4. Computer Program Listing

```
        *********ADVANCED  CONTINUOUS  SIMULATION  LANGUAGE********
      ACSL TRANSLATOR VERSION 4 LEVEL 8L1 80/134   12.01.02          I

 "TEST CASE WITH CONVERSION TO ASCL"
 "                                      "
 CINTERVAL CINT = .1
 MAXTERVAL MAXT = .010
 "                                      "
 NSTEPS NSTP = 1
PROGRAM WATERBRAKE (HPX) CONTROLLER
 "                                      "
"TABLE OF VALUE POSITION (INPUT) VS VALVE FLUID FLOW, FY1(POSIT) "
TABLE FY1,1,6                                                  ...
      /0.0,10.0,30.,60.,80.0,200.                              ...
      ,0.0,.10,.30,.60,.800,2.00 /
      "            "
      "            "
 "TABLE OF ROTOR SPEED (N2,RPMS) VS CHANGE HPX, FY2(CHPX)"
TABLE FY2,1,5                                                  ...
      /3000.,5500.,8200.,10000.,12500.                         ...
      ,75.,38.,18.,12.,8.600 /
      "            "
      "            "
 "TABLE OF ROTOR SPEED (N2,RPMS) VS WATERBRAKE RESISTANCE FY3(R4)"
 "                                      "
TABLE FY3,1,5                                                  ...
      /1000.,2500.0,10000.,11800.,13000.                       ...
      ,9.1,6.3,4.0,3.0 ,2.9 /
      "SET CONSTANTS "
 "INITIAL PRESSURES & FLOWS "
 "TABLE OF ROTOR SPEED (N2,RPMS) VS DESIRE HPX  FY4(HPXD)"
 "                                      "
TABLE FY4,1,9                                                  ...
      /3500.,4000.0,4010.,5000.,5010.,6000.,6010.,7000.,10000.  ...
      ,10.,10.,20.,20.,30.,30.,50.,50.,100. /
      "SET CONSTANT"
 CONSTANT    PS=60.00,P1I=.01,P2I=.01
 CONSTANT    W0I=0.00,W1I=0.01,W2I=.010
 "                                      "
 "INITIALIZE HPX VALUES & VALVE POSITION "
 CONSTANT  XOLD=.01,CV=.01,TRS=1.0,CW=3.88E-7 ,GC=1.144       ...
              ,XD=.01,TDL= .100
 " SET RESISTANCE (RE), INDUCTANCE (LE) & CAPACITANCE (CE) "
 CONSTANT    RE2=7.610,R5=20.00
 CONSTANT    LE2=2.46
 CONSTANT    CE2=2.13E-4,  CE1=5.23E-5
 "                                      "
 " BOUND THE SYSTEM BY PHYSICAL LIMITATIONS "
 CONSTANT  PB=.50,PT=65.0,PD1B=.20,PD1T= 60.  ...
              ,PDB=-80.0,PDT=80.0 ,CB=-500.,CT=500.  ...
           ,VB=-1., VT= 98.  ,WB=.005,WT=2.30
 "                                      "
DYNAMIC
DERIVATIVE
 "                                      "
 PS= 65.
"N2=10200.                   "
```

Table 4. Concluded

```
       *********ADVANCED  CONTINUOUS  SIMULATION  LANGUAGE*:
  ACSL TRANSLATOR VERSION 4 LEVEL 8L1 90/134   12.01.02

 N2=  4000.+ 380.*RAMP(TRS)
 "                                  "
 CV= FY1(POSIT) *1.0
 CR= FY3(N2) *1.0
 PDD=  PS-P1
" SECTION A : VALVE                 "
 P1D=BOUND(PD1B,PD1T,PDD)
 W11=.95*CV*SQRT(ABS(P1D))
 W1 = BOUND(WB,WT,W11)
"   "
 DP12=(W1-W2)/CE1
 DP1=BOUND(PDB,PDT,DP12)
 P1=LIMINT(DP1,P1I,PB,PT)
"   "
 " SECTION B : TRANMISSION LINE     "
 C8= FY2(N2) *1.0
 C9= FY1(POSIT) *1.0
 DW2=((P1-P2)-(W2*ABS(W2)*RE2))/LE2
 W2=LIMINT(DW2,W2I,WB,WT)
 "                                  "
 DP23=(W2-W3)/CE2
 DP2=BOUND(PDB,PDT,DP23)
 P2=LIMINT(DP2,P2I,PB,PT)
"   "
 P21=(P2+ABS(P2)+.0001)/2.0
 "SECTION C : WATERBRAKE             "
  W33= SQRT(ABS((P21)/FY3(N2)))
  W3= BOUND(WB,WT,W33)
 "                                  "
 "SECTION D : CONTROLLER             "
 WC=(W3+ABS(W3))/2.
 HPXA=(N2*GC)**2.*WC*CW
"HPXD=  10.0 + 20.*RAMP(TRS) "
 HPXE=HPXD-HPXA
"HPXD= 100.                    "
 HPXD= FY4(N2)
 CHPX= FY2(N2) *1.0
 DWV=HPXE*CHPX *.400
 CP= BOUND(CB,CT,DWV)
 "                                  "
 "                                  "
 POS= LIMINT(CP,XOLD,VB,VT)
 POSIT= DELAY(POS,XD,TDL,1000)
END $
 CONSTANT TSTP =30.0
 TERMT(T.GE.TSTP)
END $
END $

***TRANSLATION TIME =   0.587***
```

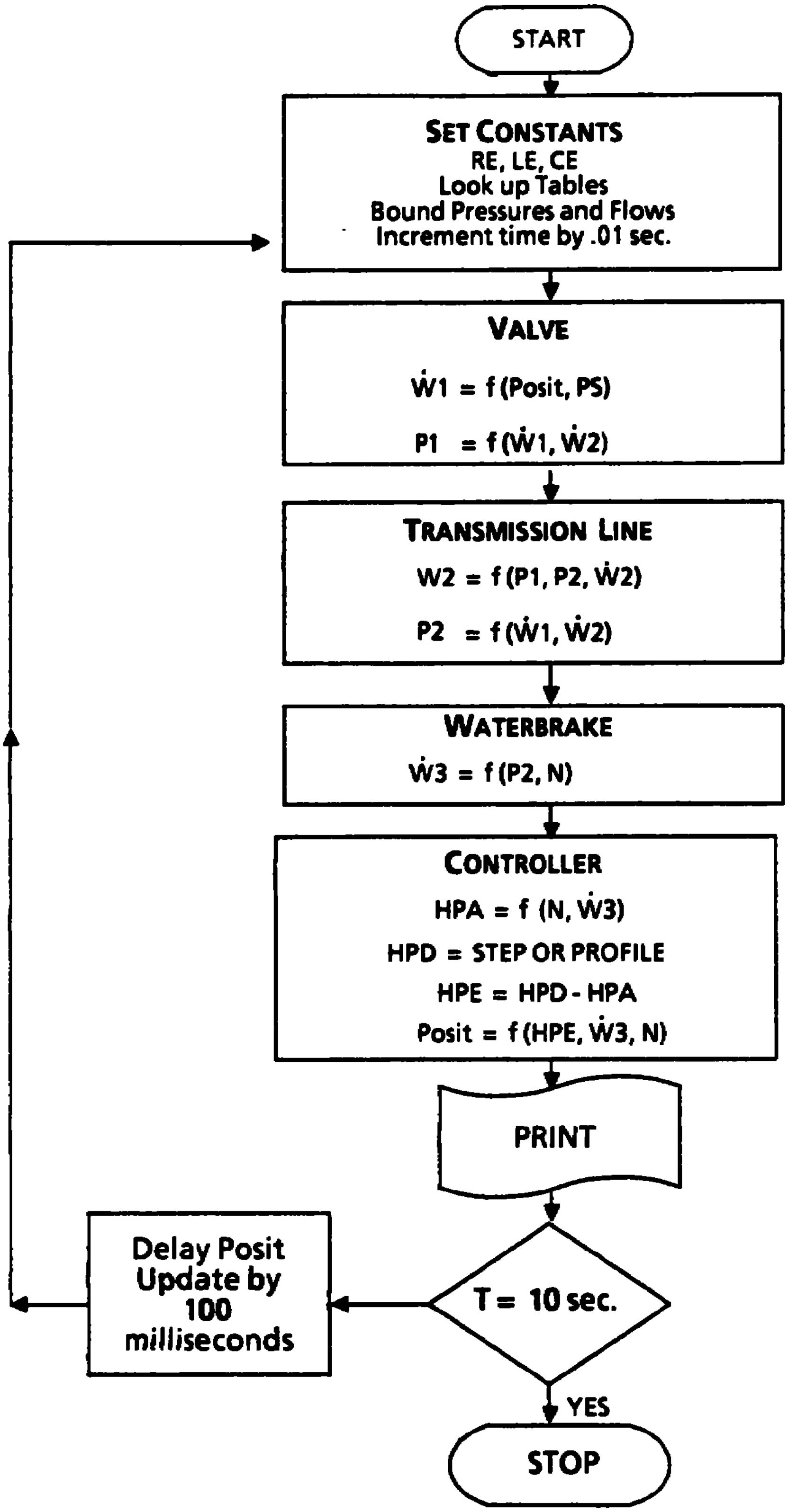


Figure 9. Computer Program Flowchart

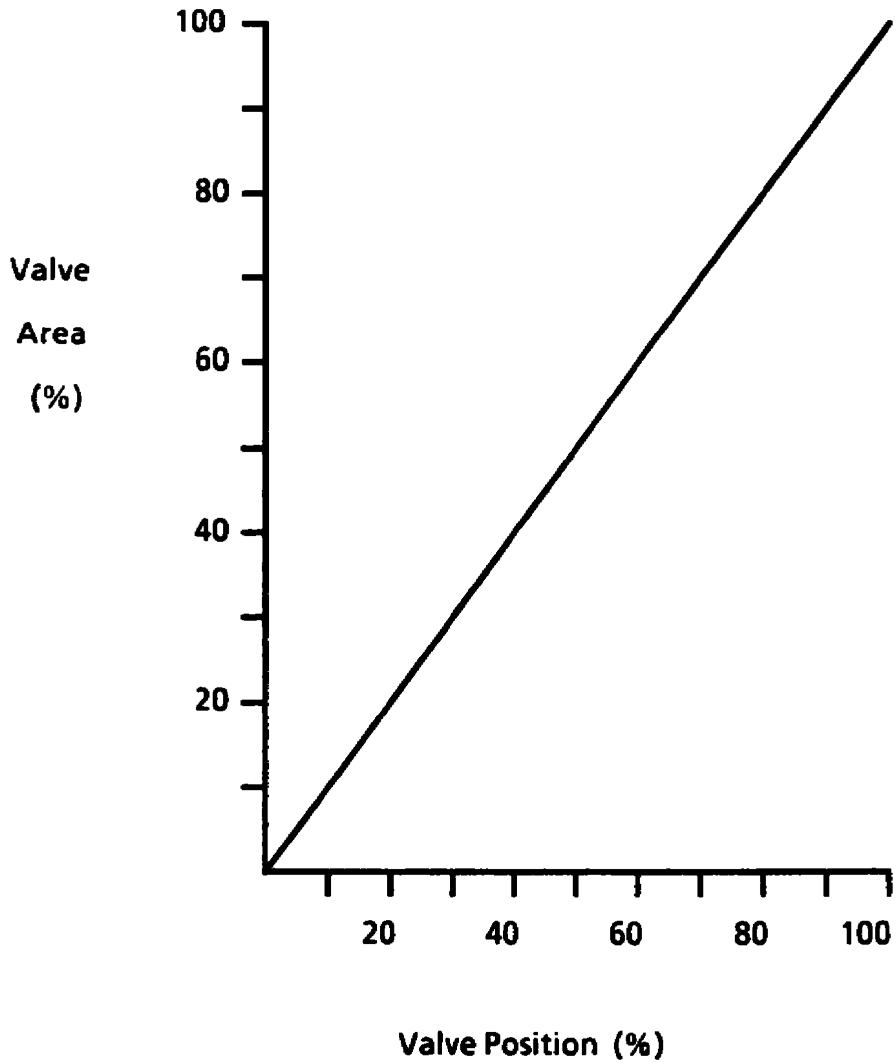


Figure 10. Plot of Valve Position to Valve Flow Area (CV)

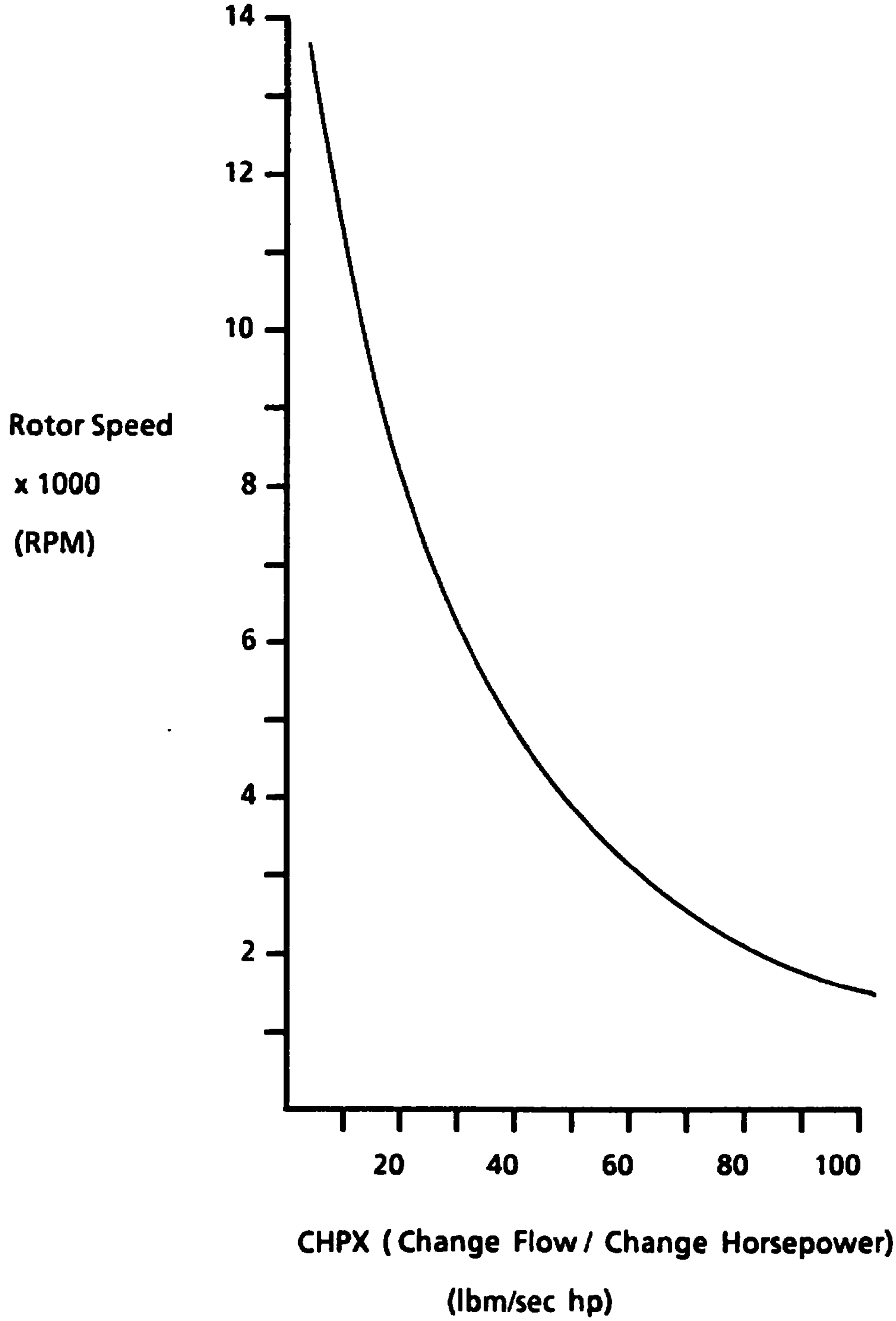


Figure 11. Rotor Speed Versus CHPX

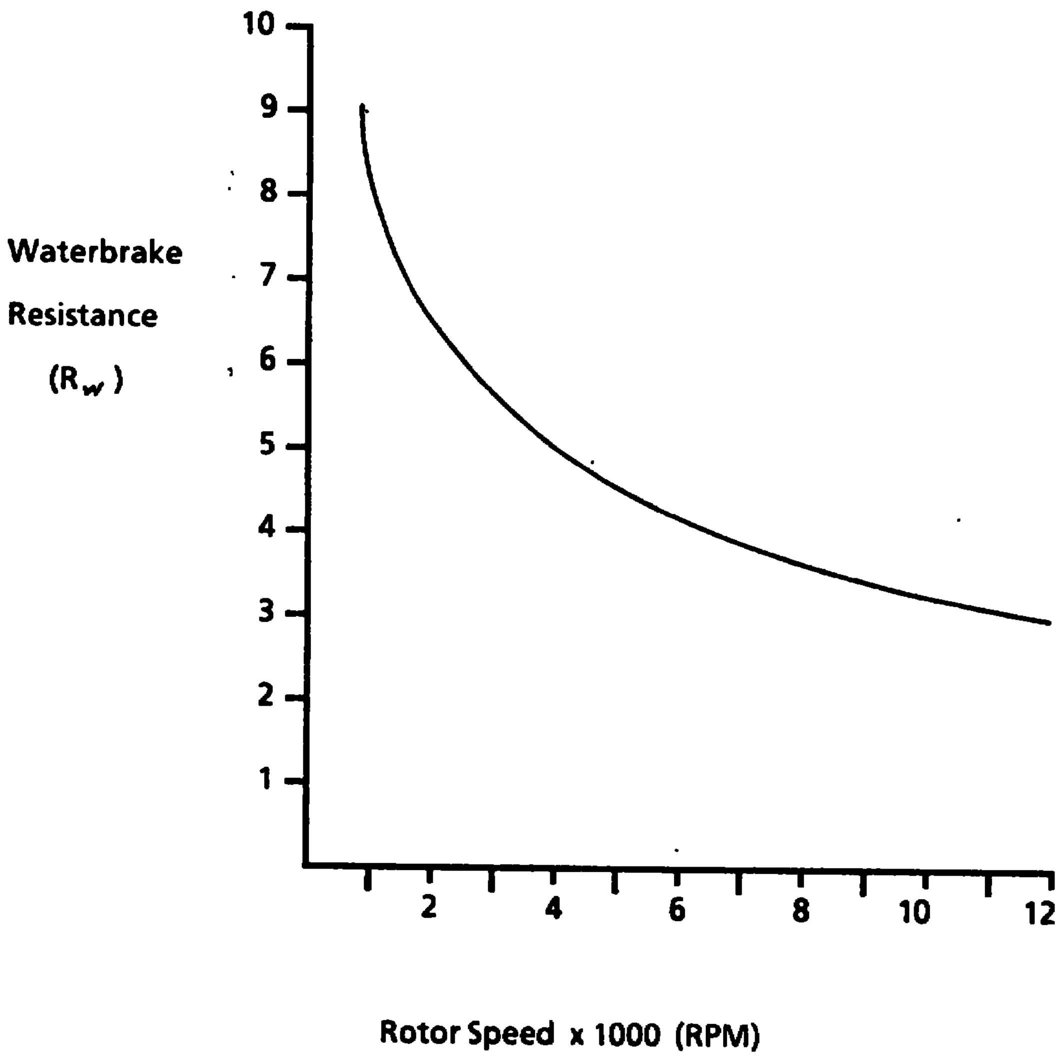


Figure 12. Rotor Speed Versus Resistance (R_W)

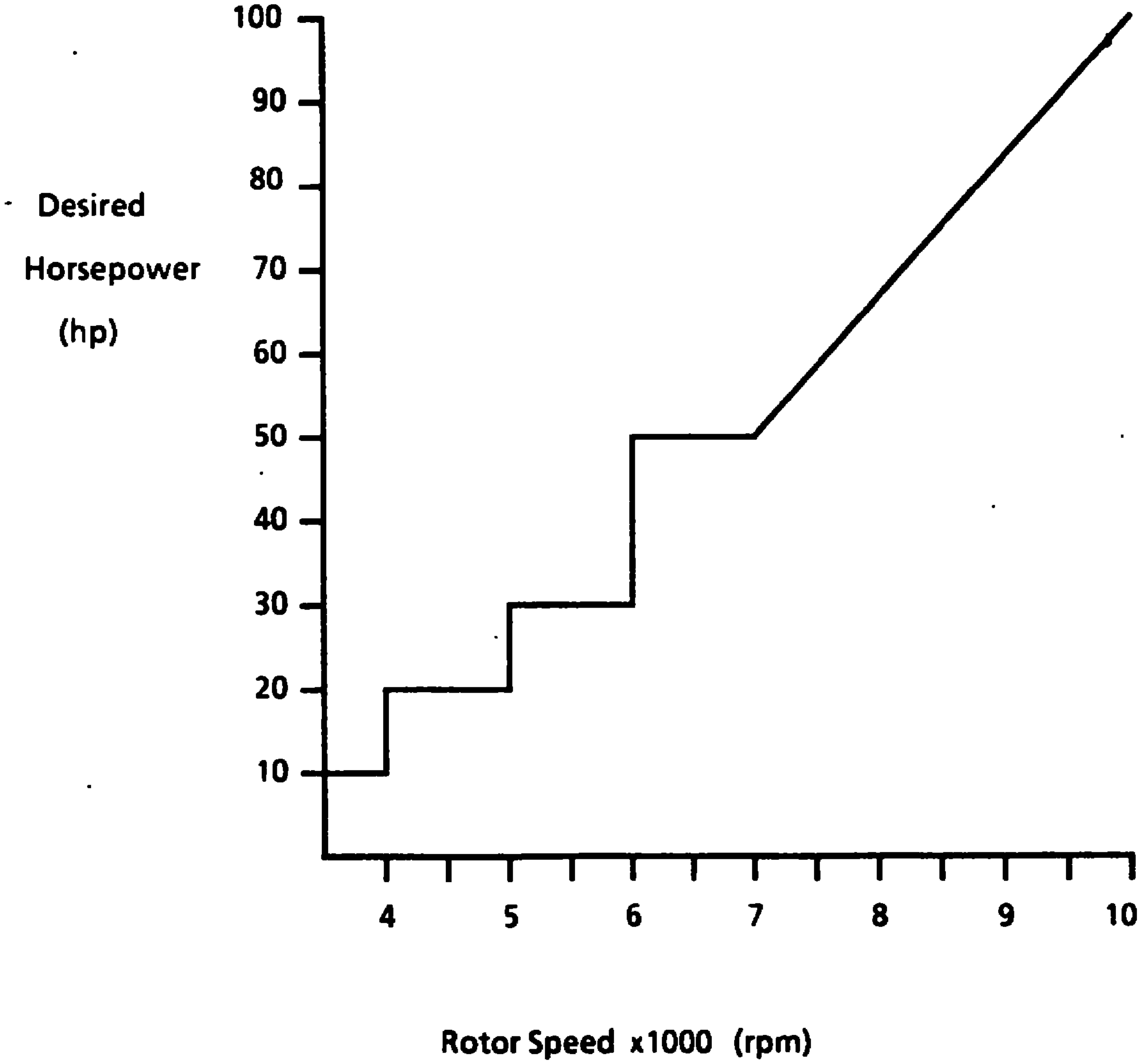


Figure 13. Rotor Speed Versus HPD

open. The update time interval of 0.1 second is within the control room's test computer capabilities. The results of various inputs will be discussed in the next chapter.

CHAPTER IV

COMPUTER MODEL RESULTS

Model Response

The computer model simulation of the horsepower Extraction System was run with step and profile input command. The step input of 100 hp command shows the steady-state accuracy. The profile input command establishes the speed of response (reaction time) of the system under transient engine operations. The results of the model simulation are validated against test data for a rotor speed increase.

Model Validation

Figure 14 compares test data to simulated data during a typical airstart rotor speed increase (360 rpm/sec). The agreement between test data and simulated data is acceptable within the range of available data (20% error at 5000 rpm) for modeling the system. The simulated data plot also shows the maximum horsepower extraction boundary for each rotor speed since the water flow is pegged at its limit of 16 gpm. Only the rotor speed will increase the amount of horsepower extracted.

Step Inputs

The response to a 100 hp, constant rotor speed set to 10,200 rpm, step input command with a one percent air entrainment is shown in Figure 15. The horsepower settles to within 99% of the command horsepower level after 2.0 seconds. The step input successfully demonstrates that accurate steady-state data (99% desired) can be obtained within a few seconds.

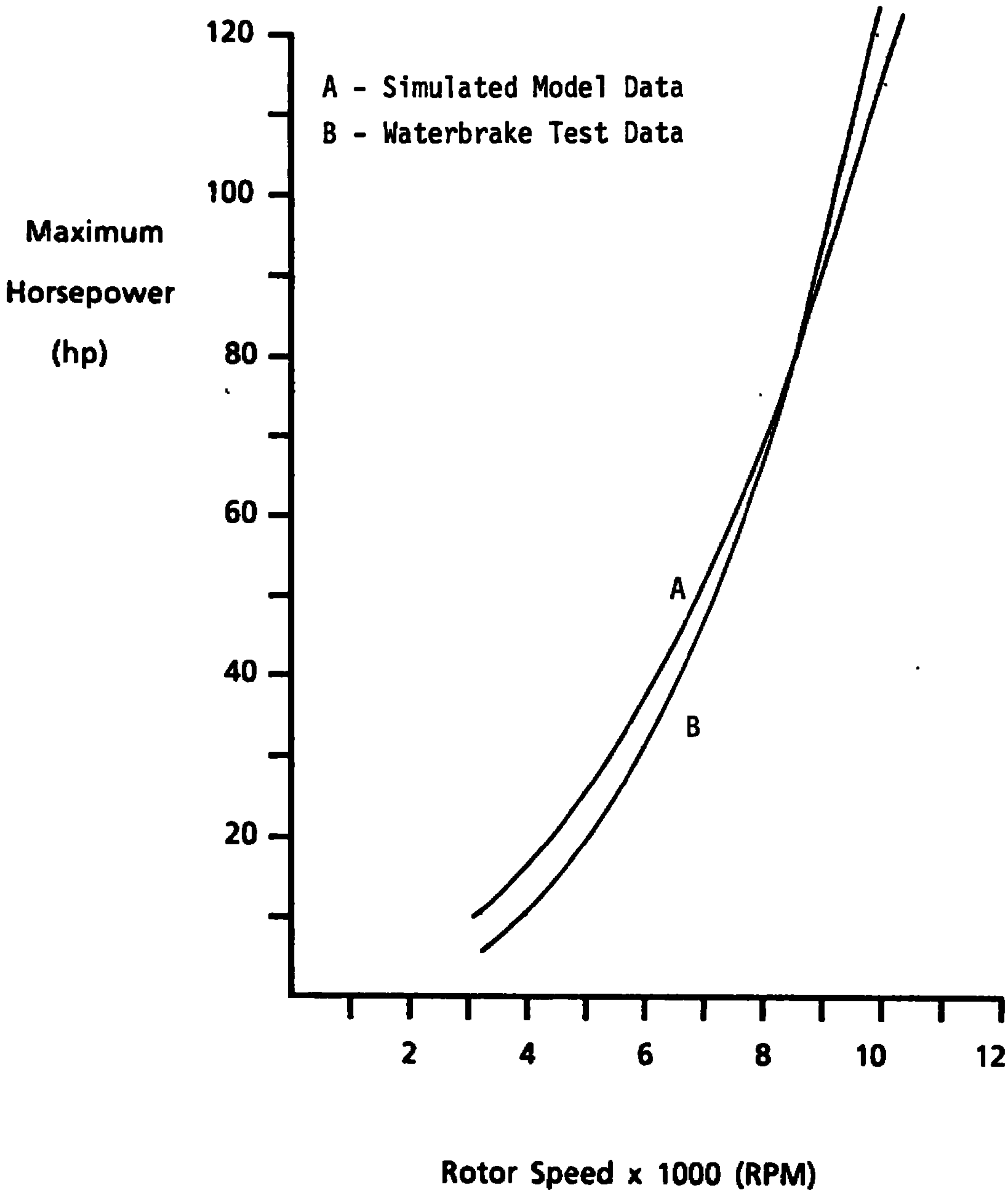


Figure 14. Comparison Between Simulated and Actual Waterbrake Data

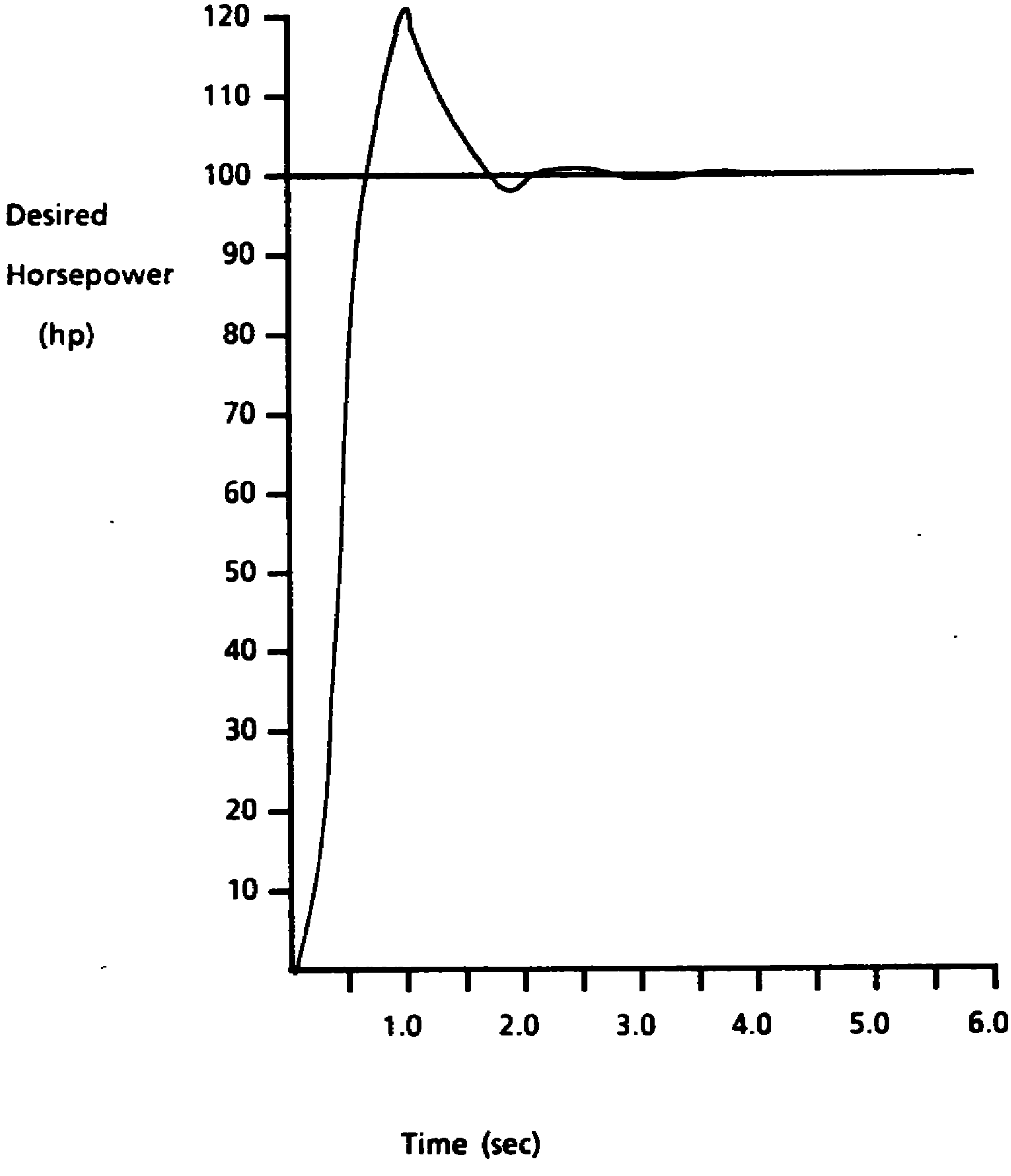


Figure 15. Response to a 100 HP Step Input Command

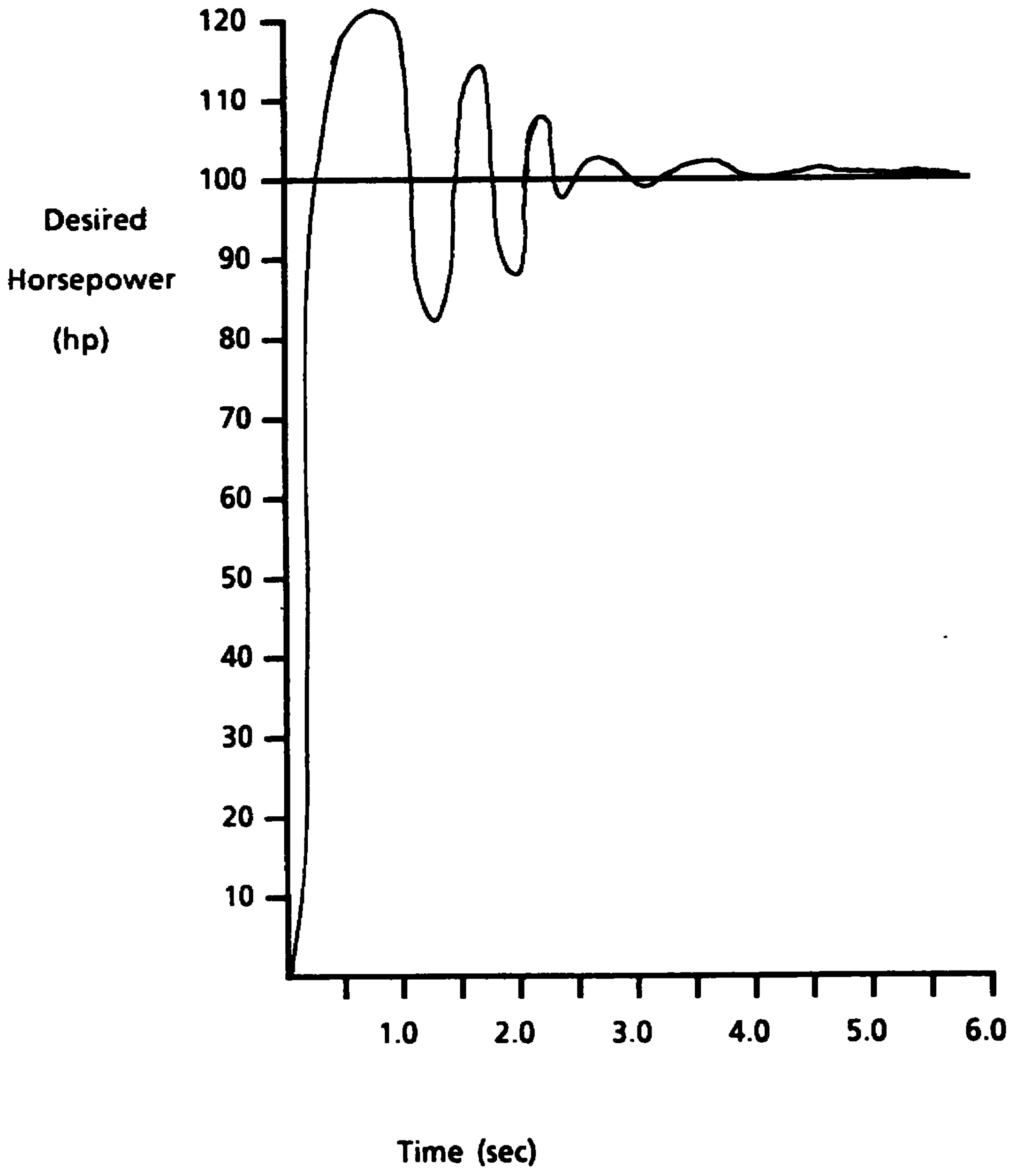


Figure 16. Response to a 100 HP Step Input Command with Reduced Length

The model response to a step input command of 100 hp with a reduced transmission line length is plotted in Figure 16. The reduced pipe length (1/10 of full length) reaches the desired horsepower approximately 0.25 seconds faster than with the normal line length but takes about three seconds longer to settle 99 percent of the desired horsepower (HPD). This short line length does not dampen out the pressure spike from the valve. The waterbrake capacitance and inductance which was not included in this model will have some effect on the results shown in Figure 16.

Profile Input

The response to profile desired horsepower (HPD) command presented in Figure 17 shows the dynamic control system response. The engine rotor speed ramped up from 4000 rpm to 10,000 rpm at a rate of 360 rpm/sec. The profile is a series of step inputs and a ramp input similar to aircraft accessory loads. The 10 hp step command increases at 4000 and at 5000 rpm are both reached after 0.9 sec overshooting the desired horsepower by approximately 3.5 hp. The step input of 20 hp at 6000 rpm reaches its maximum water flow horsepower extraction limit after 0.9 sec. The controller easily tracks a 6 hp/sec ramp command from 7000 to 10,000 rpm (at 360 rpm/sec). The control system satisfies the secondary goal of providing relative control during transient engine operations as shown in Figure 17.

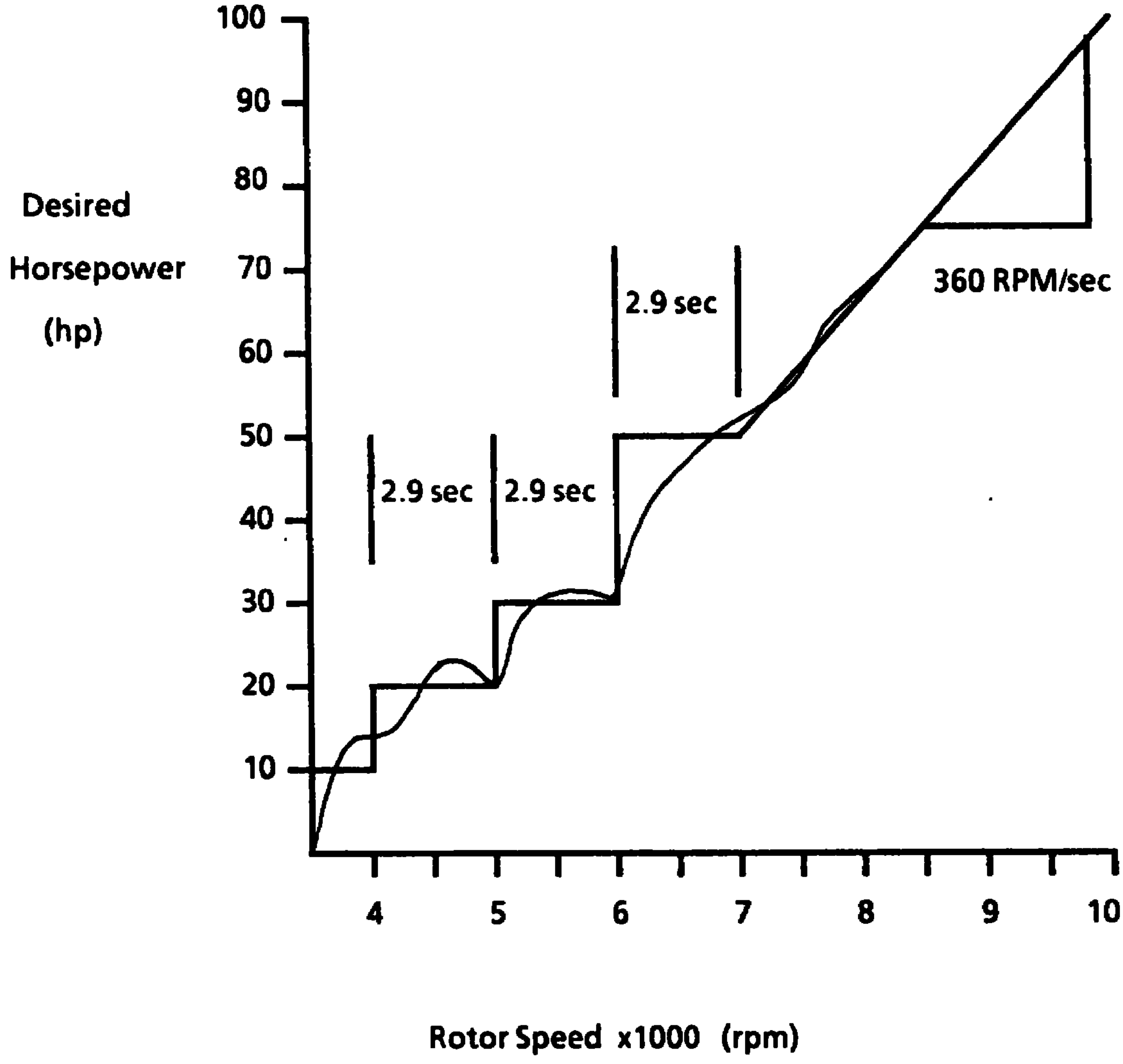


Figure 17. Response to a Profile Input Command

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

The waterbrake horsepower extraction system was modeled using the lumped-parameter technique to characterize each of the system components. A program was developed using ASCL to simulate the system response to various inputs. The results show that both of the original goals were satisfied; to obtain accurate steady-state and relatively accurate and quick transient horsepower extraction capabilities. The computer model response to a 100 hp step input shows 99% of the desired horsepower is achieved in less than three seconds versus manual control that takes about 3-5 minutes to verify steady-state horsepower level. The profile responses show the system's ability to track a series of step changes and a ramp input with much better accuracy than an operator.

It is recommended that inductance and capacitance be included in the waterbrake model to improve the system's dynamic characteristics. The inductance and capacitance terms can be quantified by measuring the waterbrake responses to known inputs.

Based on potential cost savings in turbine engine testing time, the system could recover the initial cost in a very short time. In addition, the automated horsepower extraction system would provide significant improvement in controlling horsepower during engine air starts. For the reason stated above the computer controlled extraction system would be beneficial to implement in the turbine test facilities.

LIST OF REFERENCES

LIST OF REFERENCES

1. Industrial Engineering Company, Hydrabrake (Bulletin B-12), Wendhah, New Jersey.
2. Ernest O. Doebelin, System Dynamics, Charles E. Merrill Publishing Company, Cols., Ohio, 1972.
3. Fox and McDonald, Introduction to Fluid Mechanics, John Wiley & Sons, New York, 1978
4. Baumeister, Avallone, and Baumeister, Mark's Standard Handbook For Mechanical Engineers, McGraw-Hill Book Company, New York, 1978.
5. Page S. Buckley, Techniques of Process Control, John Wiley & Sons, New York, 1964.
6. Crane Company, Flow of Fluid Through Valves, Fitting and Pipes. Chicago, Illinois, 1965.
7. The Digital Valve Company, Longmount, Colorado, 1987.
8. Mitchell and Gauther Associates Inc., Advanced Continuous Simulation (ACSL), User Guide/Reference Manual, Concord, Massachusetts, 1981.